(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 0 921 268 A3

(12) EUROPEAN PATENT APPLICATION

(88) Date of publication A3:
06.02.2002 Bulletin 2002/06

(43) Date of publication A2:
09.06.1999 Bulletin 1999/23

(21) Application number: 98203702.0

(22) Date of filing: 02.11.1998

(51) Int Cl.7: E21B 37/04, F28G 3/10, F28G 3/16

(84) Designated Contracting States:
AT BE CH CY DE DK ES FI FR GB GR IE IT LI LU MC NL PT SE
Designated Extension States:
AL LT LV MK RO SI

(30) Priority: 08.12.1997 US 67944 P
29.07.1998 US 123968

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  Designated Contracting States:
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(54) Apparatus for cleaning well tubular members

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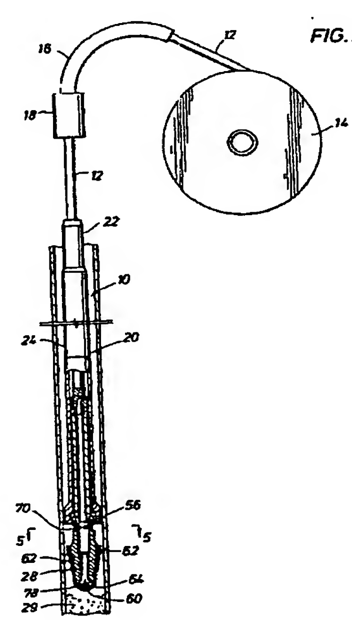

FIG.1

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European Patent Office

EUROPEAN SEARCH REPORT

Application Number
EP 98 20 3702

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
X,D	US 4 705 107 A (COUNCIL MALCOLM N ET AL) 10 November 1987 (1987-11-10) * column 4, line 10 - column 5, line 26; figures 4,5 *	1,3-6	E21B37/04 F28G3/10 F28G3/16
Y		2,3	
Y	US 3 133 603 A (JEAN-PIERRE LAPEYRE ET AL) 19 May 1964 (1964-05-19) * figure 1 *	2	
Y	US 4 909 325 A (HOPMANN MARK E) 20 March 1990 (1990-03-20) * figure 2 *	3	
A	US 4 919 204 A (BAKER WALTER ET AL) 24 April 1990 (1990-04-24) * column 5, line 37 - column 6, line 29 *	1,6	
A	FR 2 707 335 A (EPIMKIN ALEXEI ALEXEEVICH) 13 January 1995 (1995-01-13) * the whole document *	1	

TECHNICAL FIELDS SEARCHED (Int.Cl.6)

E21B
F28G
B08B

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
MUNICH	6 December 2001	Giorgini, G

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
A : technological background
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P : intermediate document

T : theory or principle underlying the invention
E : earlier patent document, but published on, or after the filing date
D : document cited in the application
L : document cited for other reasons

& : member of the same patent family, corresponding document

EPO FORM 1503 03.82 (P04C01)

ANNEX TO THE EUROPEAN SEARCH REPORT ON EUROPEAN PATENT APPLICATION NO. EP 98 20 3702

This annex lists the patent family members relating to the patent documents cited in the above-mentioned European search report.
The members are as contained in the European Patent Office EDP file on
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06-12-2001

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EPO FORM P0459

For more details about this annex : see Official Journal of the European Patent Office, No. 12/82

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(11) EP 0 921 268 A2

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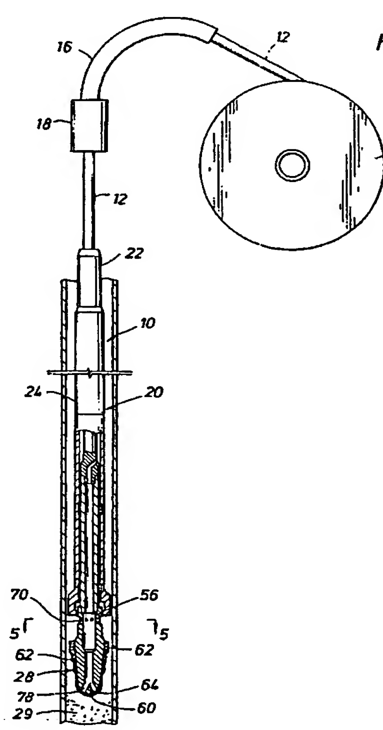
FIG.1



EP 0 921 268 A2

Description

Field of the Invention

[0001] This invention relates to apparatus for cleaning well tubular members downhole, and more particularly to a downhole cleaning tool having a rotary milling head with fluid discharge nozzles for cleaning deposits from the tubular bore.

Background of the Invention

[0002] Downhole rotating jetting heads have been used to remove deposits from well tubular members. Fluid jetting heads driven by impulse motors, such as offset nozzles or turbines, tend to generate low drive torque. This makes the heads susceptible to stalling by deposit cuttings or by deposit bridges. Further, when the head stalls, no signal is given to surface that the head is not rotating. Further, jetting heads are most efficient when cleaning radially rather than axially, particularly for drilling through deposit bridges. Therefore, multiple runs into a well are often required to fully remove the deposits; first running a positive displacement motor (PDM) with a milling head to remove bridges followed by a run with a radial jetting tool to remove wall deposits.

[0003] United States Patent no. 4,705,107 dated November 10, 1987 shows a rotating cutting tool driven by a fluid turbine motor with fluid nozzles to assist in removing deposits from the tubular bore. However, spend fluid used for driving the turbine motor is utilized for the fluid nozzles and a flexible connection is provided between the cutting head and the fluid turbine motor which substantially limits the torque transmitted to the cutter. Further, no blades or cutting elements are provided on the lower end of the cutting tool for penetrating any bridge deposits.

[0004] It is desired that a milling head rigidly connected to the motor be provided with milling elements projecting downwardly from the lower end of the head along the axis of rotation for penetrating deposits bridging the tubular bore.

[0005] Further, it is desired that fluid jets be provided adjacent the upper end of the milling elements on the milling head to assist in the cleaning of the deposits from the tubular bore after milling with unspent power fluid being available for the fluid jets.

Summary of the Invention

[0006] The present invention is directed to a downhole cleaning tool having a rotating milling head with fluid jet nozzles receiving power fluid. The power fluid is discharged radially from jet nozzles on a combined fluid jetting and milling head (jet/milling head) adjacent milling elements projecting from the head and removes deposits outside the hole drilled by the milling head. The milling elements on the milling head extend along the lower end of the milling head across the axis of rotation for drilling a hole through a deposit bridging the tubular bore and the fluid jet streams discharged from jet nozzles above the milling elements are highly effective in removing any deposits remaining after the milling action. Additional fluid jets may be provided adjacent the lower end of the milling head.

[0007] A fluid motor is utilized for rotating the jet/milling head and a major portion of the power fluid is normally directed to the fluid motor for rotating or driving the fluid motor. The remainder of the power fluid flows unimpeded to the jet nozzles for discharge radially at a relatively high velocity against the inner periphery of the adjacent tubular member. The spent fluid which was diverted to the fluid motor for rotation of the jet/milling head rejoins the power fluid downstream of the fluid motor for discharge from the jet nozzles with the power fluid.

[0008] Normally, if the jet/milling head is not stalled or milling, the pressure drop through the fluid motor is relatively low (100 psi) and the remainder of the pressure drop across the tool is across the nozzles. In the event the head begins to stall or to mill, the pressure drop across the fluid motor increases to about 400 psi, for example, providing an increase in total tool pressure drop at a constant flow rate. This gives a stall indication at surface and the flow rate may be reduced to acceptable levels. The increased pressure drop across the fluid motor develops significant torque, such as at least about 50 ft.-lbs., which should free the stuck head (followed by a pressure drop across the tool) or allow milling of a bridge.

[0009] Coiled tubing is normally used for servicing of wells to remove scale and other downhole deposits on the wells of the well tubular members. The cleaning fluid is injected in the coiled tubing and flows downwardly to the bottom hole assembly or tool which includes the fluid motor and jet/milling head. The power fluid is split between flow through the motor and flow through a bypass port through the rotor thereby giving improved speed control. The two split flows combine downstream of the motor and flow to the jetting nozzles. The amount of bypass flow is controlled by using properly sized orifices in the bypass passage to the motor. In the event the head stalls due to jamming or encountering a bridge, the available pressure drop across the motor (and therefore the torque) is limited by the pressure drop through the bypass port caused by the increased flow through the bypass port after stall.

[0010] The jet/milling head of a generally frusto-conical shape includes milling elements or inserts projecting from the outer surface of the milling head including the rounded lower end of the head and are particularly effective in breaking through a bridge across the tubular bore. Radially directed fluid discharge nozzles are positioned on the jet/milling head adjacent the upper ends of the milling elements for the radial discharge of high pressure cleaning fluid directly against the deposits in

the tubular member after the milling elements have contacted the deposits. Fluid discharge nozzles are provided on the lower rounded end of the jet/milling head for the downward discharge of high velocity fluid against the deposits prior to engagement of the deposits by the milling elements. The lower nozzles are effective also for the transport of milled cuttings upwardly above the fluid jetting and milling head.

[0011] It is an object of the invention to provide a cleaning tool for cleaning deposits from a downhole tubular member which has a combined fluid jetting and milling head with high velocity jet nozzles positioned above projecting milling elements for removing the deposits from the tubular bore of the tubular member.

[0012] Another object of the invention is to provide a cleaning tool having a lower fluid jetting and milling head with milling elements projecting from the rounded lower end of the head and fluid discharge nozzles adjacent the lower end of the milling elements for the discharge of high velocity cleaning fluid directly against the deposits prior to engagement of the deposits by the milling elements.

[0013] A further object is to provide such a cleaning tool having a fluid motor for rotating the jet/milling head with the power fluid divided between a passage for driving the motor and a bypass passage through the rotor of the fluid motor thereby permitting power or unspent fluid to flow to the jet nozzles.

[0014] Other objects, features, and advantages will be apparent from the following specification and drawings.

Brief Description of the Drawings

[0015]

Figure 1 is an elevational view, partly schematic, of the cleaning apparatus of the present invention showing a cleaning tool supported by coiled tubing downhole within a tubular member for cleaning deposits from the tubular bore;

Figure 2 is an enlarged sectional view of the upper end portion of the cleaning tool including the hydraulic fluid motor for rotating the cleaning tool;

Figure 3 is a section taken generally along the line 3-3 of Figure 2;

Figure 4 is an enlarged sectional view forming generally a continuation of Figure 2 and showing the lower end portion of the cleaning tool including a combined fluid jetting and milling head rotated by the fluid motor; and

Figure 5 is a section taken generally along the line 5-5 of Figure 4.

Description of the Invention

[0016] Referring now particularly to Figure 1, a well is illustrated having a casing 10 mounted within an earth formation. Various types of deposits may accumulate on the inner peripheral walls of the casing or tubular member 10 such as paraffin, silicates, carbonates, and sulphate, for example. Coiled tubing shown generally at 12 is normally used for servicing of wells. A reel 14 for the coiled tubing 12 stores the coiled tubing and permits unreeling of the coiled tubing 12 through a guide 16 extending to an injector 18 for inserting coiled tubing 12 downhole within the tubular member 10.

[0017] A cleaning tool generally indicated at 20 is connected by a suitable connector 22 to the lower end of coiled tubing 12. Tool 20 has an outer housing 24 and a suitable check valve (not shown) may be positioned within outer housing 24 to restrict backflow of fluids, as desired. Tool 20 includes a hydraulic fluid motor generally indicated at 26 having a shaft or mandrel 27 extending therefrom connected to a lower combined fluid jetting and milling head (jet/milling head) generally indicated at 28.

[0018] As shown in Figure 2, hydraulic fluid motor 26 has an outer stator 30 secured to housing 24 and receiving a rotor 32 having a central bore 34 therethrough and closed at its lower end at 36 by shaft 28. Central bore 34 provides a bypass fluid passage for fluid to bypass motor 26 and has an upper inlet nozzle 37 defining a bypass port or opening 38 for central bore 34. An annular flow passage 39 is provided between rotor 32 and stator 30. Rotor 32 has fins or blades extending outwardly into flow passage 39 and contacted by the downward flow of power fluid in annular passage 39 of rotor 32 thereby to rotate rotor 32 and shaft 27. Thus, the flow of fluid flowing downwardly in coiled tubing 12 is divided adjacent the upper end of rotor 32 into bypassing fluid flowing through port 38 and central bore 34, and fluid flowing downwardly in annular passage 39 outside of rotor 32 and engaging fins or blades on rotor 32 extending into passage 39 for rotation of rotor 32. The bypass fluid entering bore 34 flows outwardly from bore 34 through ports 40 into the annulus 42 between shaft 27 and housing 24 where it joins the spent fluid flowing downwardly from annular passage 39 about rotor 32. The arrangement of fluid motor 26 is shown in Figure 2 primarily schematically and various embodiments of fluid motors may be utilized in a satisfactory manner to provide bypass fluid.

[0019] Normally the fluid pressure drop through fluid motor 26 is relatively low, such as 100 psi, for example, and the remainder of the pressure drop occurs at the fluid discharge nozzles for jet/milling head 28. In the event the jet/milling head 28 stalls due to jamming or encountering a bridge across the tubular bore, a large fluid pressure drop occurs at the hydraulic motor 26 and an increased fluid flow occurs through the bypass passage 34.

[0020] It may be desirable in order to increase the torque for rotor 32 to provide an increased fluid flow to rotor 32 and a suitable valve member (not shown) responsive to a selected fluid pressure differential may be positioned within bypass port 38 and upon actuation of the valve member by an increased fluid pressure differential an increased fluid flow may be provided to annular passage 39 for rotating rotor 32. Upon an increase in the fluid pressure differential, the valve member would move to restrict the flow of fluid through bypass passage 34 thereby diverting most of the fluid through annular passage 39 outside rotor 32 for rotating rotor 32 and shaft 27 thereby to provide increased torque to fluid jetting and milling head 28. When head 28 becomes free, a decrease in circulation pressure occurs and the bypass valve member would return to its original position in which the predetermined fluid split is provided between bypass fluid moving through bypass passage 34 and power fluid for driving rotor 32.

[0021] Shaft 27 has a lower end portion 44 with a central bore 46 and fluid from annulus 42 flows through port 47 into central bore 46. Lower end portion 44 has an externally threaded lower end. Head 28 has an upper cap 48 and an internally threaded sleeve 52 extending upwardly from cap 48 is threaded onto shaft 27 for rotation therewith. Shaft 27 is mounted for relation on spaced bearings 51 between shaft 27 and outer housing 24. Upper bearing 51 blocks the downward flow of fluid in the annulus between shaft 27 and outer housing 24. Outer housing 24 has an end drift ring or hood 50 threaded thereto for receiving end cap 48 of head 28 therein. Hood 50 has an inwardly extending guide 53 adjacent shoulder 55 for contacting end cap 48 to minimize eccentric movement of head 28 during rotation thereof.

[0022] Fluid jetting and milling head 28 has a bore 56 in a tapered body extending to a rounded or hemispherical end nose 60 on the lower end of head 28. Bore 56 forms a continuation of bore 46. Fluid jetting and milling head 28 is generally frusto-conical in shape to define a tapered outer surface 59 extending upwardly from rounded end nose 60. A plurality of randomly spaced milling elements or inserts 62, preferably formed of tungsten carbide, are embedded in head 28 and project outwardly from outer surface 59 of head 28. A plurality of lowermost milling inserts 64 are embedded in rounded nose 60 adjacent the rotational axis of head 28 and project outwardly from the outer surface of nose 60 thereof to mill effectively a hole in a deposit bridging the tubular bore.

[0023] Referring to Figures 4 and 5, radially extending fluid passages 70 extend radially through jet/milling head 28 from bore 56 for a radial discharge of fluid directly against the deposit 29. While six fluid passages 70 are shown in the drawings, any desired number of fluid passages 70 may be provided, and a pair of opposed fluid passages 70 is preferred. A discharge nozzle 72 having a port or jet 74 is threaded within each of the lateral passages 70 above milling elements 62. Nozzles 72 are positioned closely adjacent drift ring 50 with the centerline of nozzles 72 preferably about ¼ inch below drift ring 50. Satisfactory results may be obtained with the centerline of ports 74 spaced vertically as much as about 2 inches from drift ring 50. Nozzles 72 are preferably spaced laterally from the inner periphery of tubular member 10 a distance between 2 and 10 times the diameter of the port 74. Thus, a spacing between about 3/8 inch to 1 ¼ inch is preferred.

[0024] A pair of lower discharge ports 78 communicating with bore 56 are provided adjacent lower milling elements 64. Ports 78 are preferably positioned at a twenty (20) degree angle to the longitudinal axis of tool 20 for discharging a fluid jet against the deposit 29 in a downward direction from milling elements 64. An angle between about ten (10) degrees and forty-five (45) degrees with respect to the longitudinal axis would function in a satisfactory manner. Suitable nozzles (not shown) may be positioned within ports 78 if desired. While two ports 78 are shown in the drawings, a single port 78 is preferable. Lower milling elements 64 extend over lower nose 60 so that direct contact is made by milling elements 64 at the center of any deposit bridge.

[0025] As an example of a satisfactory cleaning tool 20, a flow rate of 1.3 barrels per minute (bpm) was provided with two nozzles 72 having a port 74 of 0.12 inch diameter. A single lower port 78 of about 0.125 inch diameter was utilized. A normal nozzle pressure of about 1700 psi was provided for nozzles 72. The fluid motor 26 had a diameter of about 2 1/8 inches and was rotated at about 325 revolutions per minute (rpm). The diameter of head 28 was 1.50 inches and the maximum milling diameter including milling elements 62 was 1.75 inches. Ring 50 for mounting of head 28 was about 2.75 inches in diameter. A tool 20 in accord with the above was found to remove effectively soft and hard deposits from the tubular bore of tubular member 10. Head 28 as shown in the drawings is spaced a relatively small lateral distance from tubular member 10. In most instances, head 28 would be spaced a greater distance from tubular member 10.

[0026] It is apparent that various fluid nozzles may be provided above and below the milling elements 62 and 64 on fluid jetting and milling head 28. The number and port sizes of the nozzles would vary dependent primarily on the type of deposit to be removed from the tubular bore. Likewise, the amount of bypass fluid bypassing rotor 32 through rotor bore 32 would vary dependent primarily on the type of deposit to be removed. A plurality of nozzles 37 having different sizes of ports 38 may be provided with a desired port size selected for a desired amount of bypass fluid.

[0027] While preferred embodiments of the present invention have been illustrated in detail, it is apparent that modifications and adaptations of the preferred embodiments will occur to those skilled in the art.

Claims

1. A cleaning tool for removing a deposit from the bore of a tubular member having a fluid motor for rotating an output shaft and a drive shaft connected to said output shaft for rotation therewith and a combined fluid jetting and milling head operatively connected to said drive shaft for rotation and having a lower nose constructed and arranged to penetrate a deposit bridging the bore of said tubular member, said combined fluid jetting and milling head having a fluid passage in fluid communication with said fluid motor and a plurality of milling elements extending outwardly from said head including said lower nose for engaging said deposit, and a fluid discharge jet adjacent the upper end of said head in fluid communication with said fluid passage to receive fluid from said fluid motor for discharge from said jet above said milling elements for removal of said deposit from the tubular member.

2. The cleaning tool as set forth in claim 1 wherein said fluid discharge jet nozzle is provided adjacent said milling elements in fluid communication with said central bore of said rotor to receive unspent power fluid therefrom for discharge from said jet against the deposit on the tubular member.

3. The cleaning tool as set forth in claim 1 or in claim 2 wherein said combined fluid jetting and milling head has a generally frusto-conical outer surface and said lower nose has a rounded lower end surface of a generally hemispherical shape, and said milling elements are embedded in said head and project outwardly from said frusto-conical outer surface and said nose.

4. In cleaning tool as set forth in any preceding claims wherein a fluid discharge jet is provided in said nose for discharge of cleaning fluid downwardly against said deposit prior to engagement of said milling elements with said deposit.

5. A cleaning tool as set forth in any of the preceding claims wherein the fluid motor is a hydraulic :fluid motor having an inner rotor and outer stator arranged in concentric relation and defining a fluid flow passage therebetween for the downward flow of power fluid to rotate the rotor, said rotor having a central bore and an upper entrance port for the central bore for defining a fluid bypass passage to permit power fluid to bypass the motor;

6. An Apparatus for removing deposits from a downhole tubular member comprising

   - a cleaning tool according to any of claims 1 to 5, positioned downhole in the tubular member adjacent a deposit to be removed and
   - a coiled tubing string extending from a surface location having a lower end connected to the cleaning tool and supporting the cleaning tool for movement along the tubular member

7. The apparatus as set forth in claim 6 wherein said tool has an outer housing in which said drive shaft and said head are mounted for relative rotation, and a hood is mounted on the lower end of said housing; said head having an upper end portion received within said hood and contacting said hood to minimize relative eccentric rotation of said head.

8. The apparatus as set forth in claim 7 wherein said fluid discharge jet is mounted on said upper end portion of said head below said hood.

9. The apparatus as set forth in claim 7 wherein a pair of radially directed opposed fluid jets are mounted on said upper end portion of said head below said hood and above said milling elements.

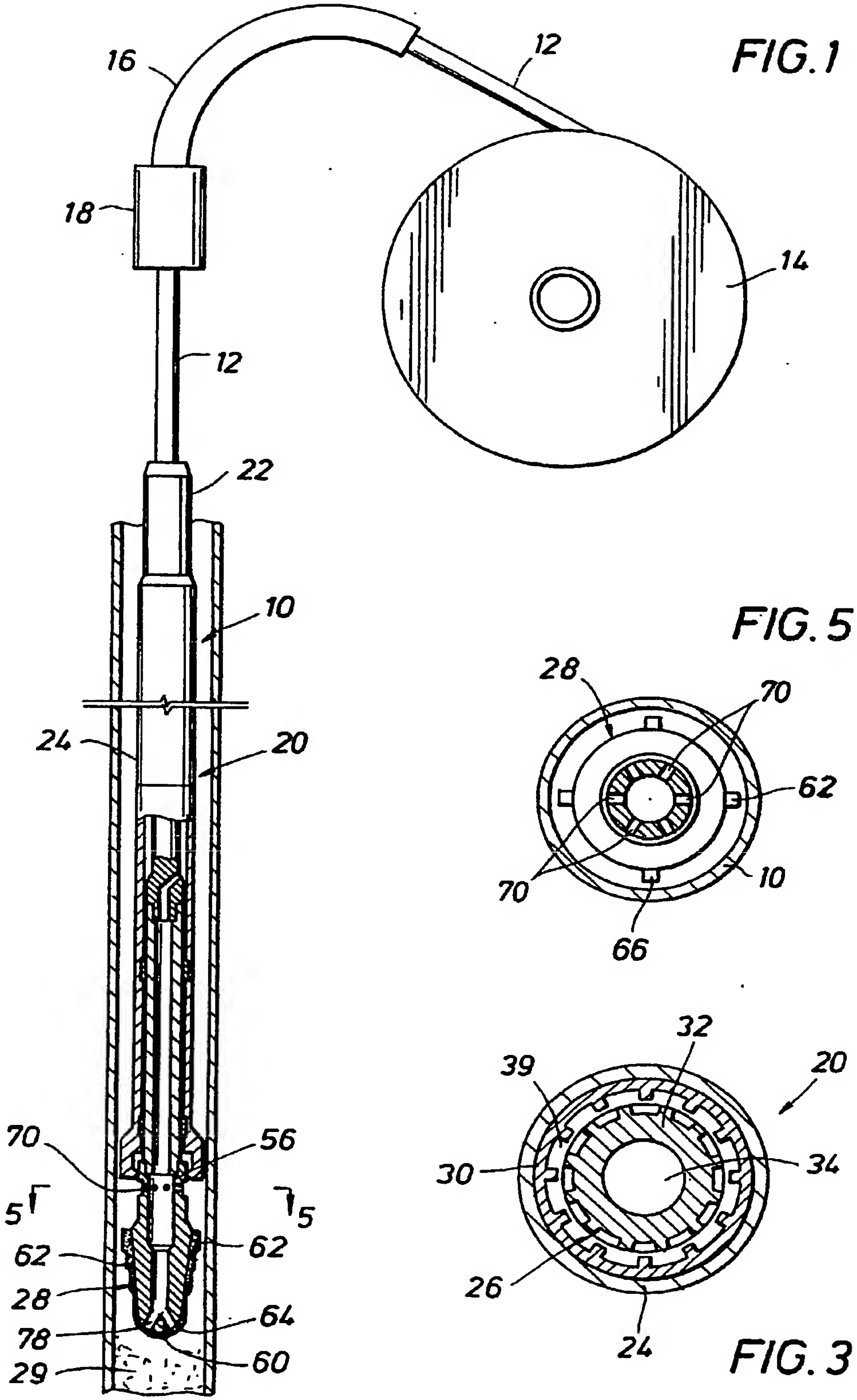


FIG. 1

FIG. 5

FIG. 3

FIG. 2

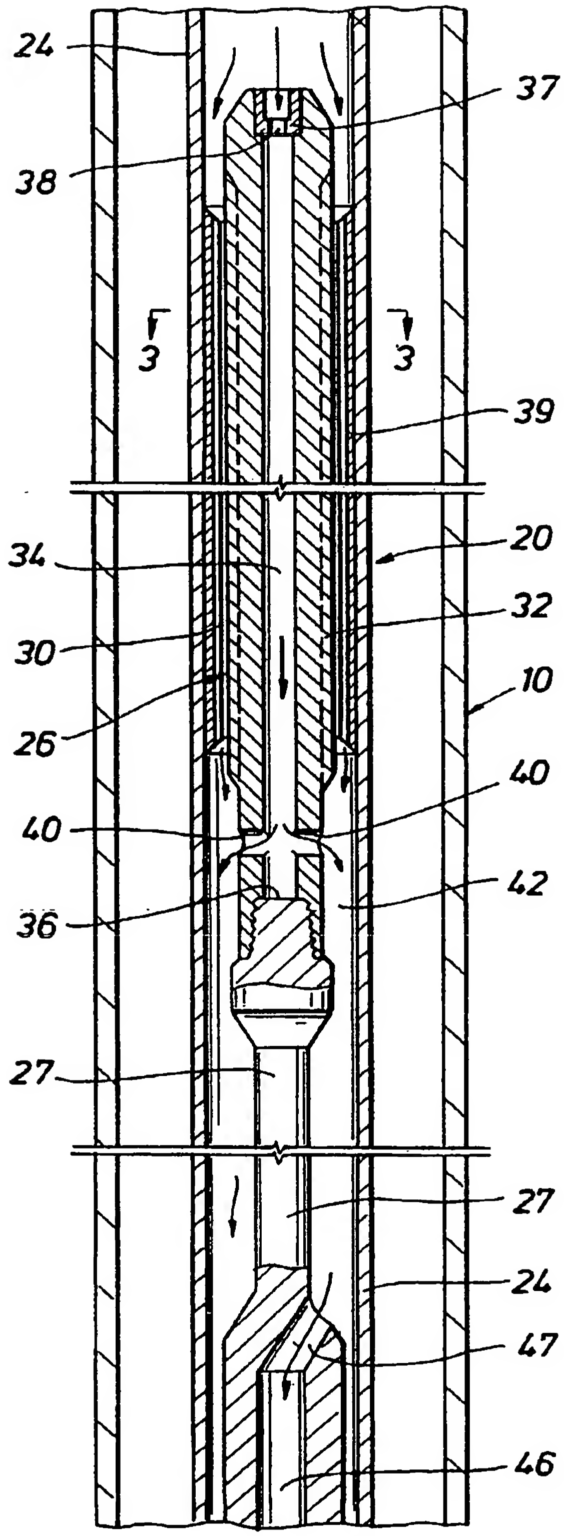

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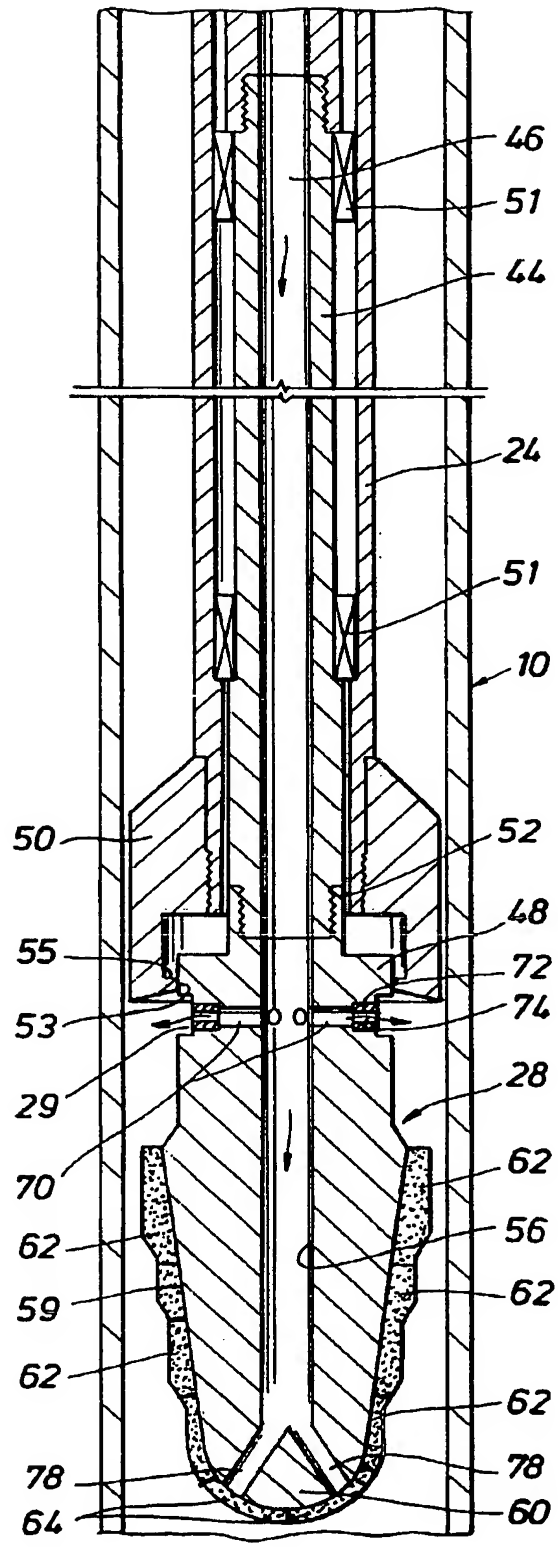
FIG. 4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PATENT SPECIFICATION

Inventor: SIR FRANK WHITTLE

762,749

Date of filing Complete Specification Sept. 26, 1955.

Application Date Dec. 10, 1954. *No. 35905/54.*

Complete Specification Published Dec. 5, 1956.

Index at acceptance :—**Classes 85,** A1D2; **110(3),** H2(DX : H3); **and 135,** P(1F : 16E3 : 24KX).

COMPLETE SPECIFICATION

Improvements in or relating to Well Drilling Systems and methods of Operating such Systems

We, N. V. DE BATAAFSCHE PETROLEUM MAATSCHAPPIJ, of 30 Carel van Bylandtlaan, The Hague, The Netherlands, a company organised under the laws of the Netherlands, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The present invention relates to well drilling systems as may be used for example in drilling deep bore holes in the surface of the earth with the object of finding salt, sulphur, water, oil or the like.

The invention particularly relates to well drilling systems of the type in which a tubular drill string, which in operation extends from the top of the bore hole to the bottom, is provided at its lower end with a hydraulic turbine for driving a rotary drill bit. The turbine is driven by fluid pumped down through the drill string, the fluid usually being what is known as a mud flush. Usually, too, the fluid is ejected through jet openings in the rotary drill bit, and is directed on to the cutting edges to cool them and to remove the cuttings, before circulating back to the surface.

When drilling with such a system through formations of varying hardness, the turbine speed will vary between wide limits, as a result of the varying resistance met by the bit. This may lead to stalling of the turbine when drilling in soft formations, or overspeeding in hard formations. In any case excessive deviations from the design speed of a turbine leads to inefficient operation. The turbine speed can be controlled manually by the driller at the top of the bore hole by varying the weight on the bit and/or the circulation rate of the mud flush which is supplied through the drill string to the turbine, in accordance with information, which is obtained about the turbine operation, such as the turbine speed or the pressure drop across the turbine. It is, however, often very difficult to communicate these measured variables from the bottom of the bore hole to the surface.

These difficulties may be partly overcome by employing a system for example that described in our Patent Application No. 35904/54 (Serial No. 755,207) with reference to Figures 2 to 6 of the drawings, in which a ram device is interposed with the hydraulic turbine between the drill string and the rotary drill bit.

The ram device described comprises two hollow telescopic members, the inner one of which provides a channel of the passage of fluid from the drill string to the bit and which members have means for preventing relative rotational movement between them, one of said members being provided with one or more pistons, which co-operate with an equal number of cylinders formed on the other member, each piston dividing the cylinder with which it co-operates into two cylindrical spaces, the ram device also comprising means for connecting the cylindrical spaces located to one side of the pistons to a space in which there is a relatively high fluid pressure in operation, and means for connecting the cylindrical spaces located to the other side of the pistons to a space in which there is a relatively lower fluid pressure in operation. Such ram devices will be referred to in this specification as "ram devices of the type specified".

In the simplest case, the space in which there is a relatively high fluid pressure is the interior of the inner telescopic member, and the space in which there is a lower fluid pressure is the exterior of the outer telescopic member (i.e. the borehole) so that the fluid pressures are those of the mud flush in the drill string and in the borehole respectively. The pressure difference applied to the pistons, and thus controlling the bit weight, is the difference of these two mud flush pressures, which is equal to the sum of the pressure drops across the turbine and the bit jet openings, both of which depend on the mud flush circulation rate. As explained in the aforesaid patent specification the bit

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Price 25p

weight determines the load on the turbine for a given hardness of the formation being drilled, and there is thus an interaction between the ram device and the turbine, which leads to an inter-relationship between the formation hardness, the turbine speed and the mud flush circulation rate. Here again it is important to restrict the variations in turbine speed in operation.

One may define in this connection, the insensibility of a system as being the ratio between a range of formation hardness, and the corresponding turbine speed range when drilling through the said range of formation hardness. Thus, the larger the insensibility of a system, the greater the range of formation hardness which may be drilled through without the turbine speed passing outside predetermined limits.

Whilst a system including a ram device represents an improvement on a system without such device, further measures to increase the insensibility are desirable.

It is an object of the present invention to provide a method and means for increasing the insensibility of a well-drilling systems which include a hydraulic turbine either alone or in combination with a ram device as described in the said patent application.

According to the present invention in operating a well-drilling system provided with a hydraulic turbine, or a combination of a hydraulic turbine and a ram device of the type specified, interposed between the drill string and a rotary drill bit, means being provided for circulating fluid through the drill string to the turbine, the circulation rate through the turbine is varied to keep the turbine speed within restricted limits, by passing part of the fluid through a channel by-passing the turbine.

The passage of fluid through the channel by-passing the turbine may be controlled by a valve responsive either to the pressure drop across the turbine or to the speed of rotation of the turbine. In either case the pressure drop across the turbine is preferably not permitted to rise above a predetermined value.

According to another aspect of the present invention a well drilling system provided with a hydraulic turbine, or a combination of a hydraulic turbine and a ram device of the type specified, for location near the lower end of a drill string, the turbine being arranged to drive a rotary drill bit and means being provided for circulating fluid through the drill string to the turbine, also includes a channel by-passing the turbine, and means for automatically controlling the flow of fluid through the channel to keep the turbine speed within restricted limits in operation when the fluid is pumped through the drill string at a constant rate.

Said means for controlling the flow of fluid through the channel by-passing the turbine may comprise a valve located in the channel, which valve is responsive either to the pressure drop across the turbine or to the speed of rotation of the turbine. In either case the valve preferably operates to prevent the pressure drop across the turbine rising above at a predetermined value.

Where the valve is responsive to the pressure drop across the turbine, it may comprise a valve member which is urged towards its closed position by a spring, and which in operation is subjected on one side to fluid at the pressure of the fluid at the turbine inlet and on the other side of fluid at the pressure of the fluid at the turbine outlet, in such a manner that the pressure difference acts in opposition to the spring.

Where the value is responsive to the speed of the turbine it may comprise a valve member urged towards its closed position by a spring, and a centrifugal governor, which, as the speed of rotation of the turbine increases, causes a force to be exerted on the valve member in opposition to the spring. The centrifugal governor may for example control the rate of supply of fluid under pressure to a servomechanism, which, as the said rate increases, exerts an increasing force on the valve member in opposition to the spring.

In any of the above cases, the valve may be a sleeve valve.

The channel by-passing the turbine may comprise a channel through the turbine rotor shaft, which is thus hollow, the channel communicating at one end with the turbine inlet and at the other end with the turbine outlet, and containing the valve or other means for controlling the flow of fluid through it.

Further according to the present invention, there is provided a hydraulic turbine for location near the end of a drill string, the turbine being adapted for driving a rotary drill bit and itself being capable of being driven by a supply of fluid pumped down through the drill string, the turbine being provided with a channel by-passing its blading, and means for automatically controlling the flow of fluid through the channel, so as to maintain the speed of rotation of the turbine within restricted limits in operation, when the fluid is pumped through the drill string at a constant rate. The said channel may pass through the turbine rotor shaft communicating at one end with the turbine inlet and at the other end with the turbine outlet.

Examples of well drilling systems in accordance with the present invention will now be described with reference to the Figures 1—4 and 6 of the drawings accompanying the provisional specification in which

Figure 1 is a cross-sectional view of the ram device/hydraulic turbine arrangement, comprising the by-pass valve as shown in detail in Figure 2,

Figure 3 shows the turbine pressure drop/turbine speed (Δp/N) curves for various

values of the circulation Q, of the fluid passing through the turbine blading,

Figure 4 shows the torque/turbine speed (T/N) curves for various values of circulation rate Q, of the fluid passing through the turbine blading, and

Figure 6 shows servo-motor operated by-pass valve.

Referring first to Figure 1, the turbine 1 comprises a stator housing 2 and a hollow rotor shaft 3, on the two of which the turbine blading 4 is mounted. The rotor shaft 3 is supported by bearings 5 and 6, whilst the rotary drill bit 7, which may be of any suitable type, is connected to the lower end of the shaft 3.

Between the drill string 8 (which is kept stationary or is rotated only very slowly so as to present it from sticking) and the hydraulic turbine 1, there is interposed a ram device as described in our Patent Application No. 35904/54. (Serial No. 755,207).

The ram device consists of a number of piston and cylinder combinations (for the sake of simplicity only two of these combinations are shown in the drawing but there may be more), the pistons 9 being connected to the inner hollow cylindrical member 10 and the cylinders 11 being formed on the outer hollow cylindrical member 12. Suitable sealing means (not shown) are provided between the pistons 9 and the interior surface of the outer member 12, and for sealing adjacent cylinders 11 from one another. The members 10 and 12 are telescopically arranged, a splined coupling 13 being provided to prevent any relative rotational movement between them. The cylindrical spaces located above each of the pistons 9 are connected to the interior of the inner telescopic member 10 by the ports 14 provided in the wall of the member 10, the ports 14 each being located just above one of the pistons 9, whilst the cylindrical spaces located below each of the pistons 9 are connected to the exterior of the outer telescopic member 12 by the ports 15 provided in the wall of the member 12, the ports 15 each being located at the lower end of the one of the cylinders 11 with which they are associated.

It will be clear that the difference existing between the pressure of the mud flush passing through the inner telescopic member 10, and the pressure of the mud flush in the borehole at the outside of the outer telescopic member 12 will be applied to all the pistons 9 and will exert on them a downwards force, which, in combination with the weight of the inner telescopic member 10, the turbine 1 and the bit 7, and the hydrodynamic force generated by the mud flush flowing through the turbine 1 and the bit 7, will constitute the effective bit weight, i.e. the load on the bit.

Inside the hollow rotor shaft 3, there is provided a pressure-sensitive valve 16, which is urged towards the closed position by a compressed spring 17. The valve 16 will be closed for all conditions under which the pressure drop across the turbine 1 is below a predetermined value, and, whilst it is closed, the mud flush emerging from the inner telescopic member 10 will flow to the jet openings of the bit 7 through the turbine blading 4 and the ports 18 provided in the wall of the rotor shaft 3. When the pressure drop across the turbine rises above said predetermined value, the valve 16 will be opened, and part of the mud flush will by-pass the turbine blading 4 by flowing through the ports 19, the inside of the rotor shaft 3 and the valve 16. Thus, when the valve 16 is open, the pressure drop across the turbine 1 will remain substantially constant, any increase in the rate of flow of mud flush down the drill string 8 resulting mainly in an increased flow through the by-pass channel.

It will be appreciated that in an alternative arrangement the valve 16 controlling the by-pass channel may be constituted by a piston-operated sleeve valve, one side of the piston being exposed to the pressure of the mud flush at the inlet of the turbine 3, and the other side to the pressure of the mud flush at the outlet of the turbine 3.

The relation between the turbine pressure drop $\triangle p$, the turbine speed N and the circulation rate Q of the mud flush passing through the turbine blading 4 is shown by the curves in Figure 3. The turbine pressure drop/speed ($\triangle p$/N) curves are shown for various circulation rates Q. As indicated in the drawing, the circulation rate Q_1 is greater than Q_2, which in turn is greater than Q_3. The valve 16 is set to operate when the pressure difference across it is $\triangle p_c$, so that when starting to operate the turbine 3 with a constant mud flush circulation rate Q_1 through the drill string 8, the turbine speed increases from zero up to a speed N_1 at which the pressure drop across it has increased to $\triangle p_c$ and the valve 16 is opened. At speeds higher than N_1, the pressure drop across the turbine will remain substantially equal to $\triangle p_c$, the circulation rate through the turbine 3 however continuously dropping as the speed increases passing through the value Q_2 at speed N_2, the circulation rate through the by-pass valve 16 then being (Q_1—Q_2). Similarly at the turbine speed N_3, the corresponding circulation rates are Q_3 and (Q_1—Q_3).

The variations in the torque delivered by the turbine, as a result of by-passing part of the mud flush, may be seen in Figure 4, which shows the torque/speed (T/N) curves for the various circulation rates Q_1, Q_2 and Q_3 of the mud flush passing through the turbine blading 4.

It will be appreciated that when starting from the point 0 on the curve Q_1, the torque will be equal to T_1 when the turbine speed has increased to N_1. At this moment (see Figure 3) the pressure drop across the

turbine is equal to $\triangle p_c$ and the valve 16 (see Figures 1 and 2) opens. At the speed N_2, the circulation rate through the turbine will be Q_2 (Figure 3), which corresponds to a torque T_2, whilst at the speed N_3 the respective values are T_3 and Q_3.

It will be clear that, although only one curve (in this case Q_2) is shown as being representative of the circulation rates lying between the values Q_1 and Q_3, it is possible to draw intermediate curves for circulation rates lying between Q_1 and Q_3, in order to be able to plot the curve $\triangle p_0$ in Figure 4 more exactly. Figure 4 shows very clearly the effect of the invention, namely that the by-pass valve 16 operates as a means of achieving a rapid rate of change of torque with speed. It will be clear that the greater this rate the greater is the insensibility of the system.

The object of the invention may be attained in another manner by inserting a speed-sensitive valve in the by-pass of the turbine. The valve which is operated by a centrifugal governor is set in such a way that starting from the speed N_1, an increasing quantity of mud flush is by-passed at increasing turbine speeds. It will be appreciated that this alternative arrangement will also result in the rate of change of torque with turbine speed being much greater than would be obtained with a turbine operating without a by-pass. However, with the speed-sensitive valve the pressure drop across the turbine can be made to decrease as the speed increases, as a result of which the torque/speed curve obtained is even steeper than that shown in Figure 4, thus giving an even greater insensibility.

In a preferred arrangement, the speed-sensitive by-pass valve is constituted by a servo-operated valve, the pressure fluid (mud) supply to the servomechanism being controlled by a centrifugal governor.

Such a bypass valve is shown in Figure 6, mounted in the interior of the hollow rotor shaft 3. The valve body 16, which is of the mushroom type (but which may also be of the sleeve type) is connected by a hollow stem 21 to the servo piston 22 which is arranged to slide in the servo cylinder 23. The valve body 16 is urged towards its closed position by the compressed spring 20. The cylindrical space below the piston 22 is connected to the turbine outlet space through the ports 27 and the interior of the hollow stem 21. A small bleeding duct 26 is provided through the piston 22. In the head of the cylinder 23, there is mounted a pilot valve 25 which is controlled by a centrifugal governor 24.

When the pilot valve 25 is fully closed at low speeds, the pressures in the spaces above and below the servo-piston 22 are equalised, owing to the presence of the duct 26, at a value equal to the fluid pressure at the turbine outlet.

When the turbine speed increases, the centrifugal governor 24 opens the pilot valve 25 and fluid from the space inside the shaft 3 enters the space in the servo cylinder 23 above the piston 22. Consequently the pressure in the space above the servo-piston 22 will rise until the rate of escape of fluid through the duct 26 is equal to the rate at which it enters through the pilot valve 25.

The force exerted by the pressure difference on the two sides of the piston 22 will displace the valve body 16 until the force on the piston 22 is in equilibrium with the opposing force due to the compression of the spring 20. The valve will then pass part of the mud flush, which would otherwise flow through the turbine blading 4.

The by-pass valve and the associated mechanism may be so designed that by their action the pressure drop across the turbine is kept substantially constant over the speed range in which it operates, thus giving a torque-speed curve similar to that shown in Figure 4. If desired, however, the by-pass valve and the associated mechanism may be designed, as already mentioned, so that the pressure drop across the turbine decreases with increasing speed, thus giving a steeper torque-speed curve than that shown in Figure 4.

It will be appreciated that the pressure drop prevailing across the valve itself may be kept to a minimum by using a sleeve valve.

Further, the by-pass valve may be located above the turbine blading. In such a case the rotor shaft need not be hollow for the fluid passing through the valve may be diverted into the borehole, thus by-passing both the blading and the rotary drill bit, or it may flow to the bit through a passage provided through or around the stator housing.

In general it is desirable to coat all the surfaces of the by-pass valve which are subjected to the erosive action of the mud flow with wear-resisting materials, and to design the by-pass valve in such a way that it is readily replaceable.

What we claim is:—

1. A well drilling system which is provided with a hydraulic turbine, or a combination of a hydraulic turbine and a ram device of the type specified, for location near the lower end of a drill string, the turbine being arranged to drive a rotary drill bit and means being provided for circulating fluid through the drill string to the turbine, and also includes a channel by-passing the turbine, and means for automatically controlling the flow of fluid through the channel to keep the turbine speed within restricted limits in operation when the fluid is pumped through the drill string at a constant rate.

2. A well drilling system according to Claim 1 in which the channel by-passing the turbine is a channel through the turbine rotor shaft, communicating at one end with the turbine

inlet and at the other end with the turbine outlet.

3. A well drilling system according to either Claim 1 or Claim 2, in which said means for controlling the flow of fluid through the channel by-passing the turbine comprises a valve located in the channel.

4. A well drilling system according to Claim 3 in which the valve is adapted to respond in operation to the pressure drop across the turbine, to open when the pressure drop rises to a predetermined value.

5. A well drilling system according to Claim 4 in which the valve comprises a valve member which is urged towards its closed position by a spring, and which in operation is subjected on one side to fluid at the pressure of the fluid at the turbine inlet and on the other side to fluid at the pressure of the fluid at the turbine outlet, in such a manner that the pressure difference acts in opposition to the spring.

6. A well drilling system according to either one of Claims 4 and 5 in which the valve is adapted to operate to prevent the pressure drop rising above said predetermined value.

7. A well drilling system according to Claim 3, in which the valve is adapted to respond in operation to the speed of rotation of the turbine, to open when the speed rises to a predetermined value.

8. A well drilling system according to Claim 7 in which the valve comprises a valve member urged towards its closed position by a spring, and a centrifugal governor mounted on the rotor, which, as the speed of rotation of the turbine increases, causes a force to be exerted on the valve member in oppositon to the spring.

9. A well drilling system according to Claim 8 in which the centrifugal governor is arranged to control the rate of supply of fluid under pressure to a servo-mechanism, which, as the said rate increases, exerts an increasing force on the valve member in opposition to the spring.

10. A well drilling system acocrding to any one of Claims 7—9 in which the valve is adapted to operate to prevent the speed rising above said predetermined value.

11. A well drilling system according to Claim 10 in which the valve is adapted to operate to cause the speed to decrease progressively from said predetermined value, as the fluid circulation rate increases beyond the value at which the speed just rises to the predetermined value.

12. A well drilling system according to any one of Claims 3—11 in which the valve is a sleeve valve.

13. A well drilling system according to Claim 1 in which the channel by-passing the turbine communicates at one end with the circulating fluid channel upstream of the turbine, and at the other end with the space in the borehole outside the drill string and the turbine.

14. A hydraulic turbine for use in a well drilling system according to any one of Claims 1—12 for location near the end of a drill string, the turbine being adapted for driving a rotary drill bit, and itself being capable of being driven by a supply of fluid pumped down through the drill string, the turbine being provided with a channel by-passing its blading, and means for automatically controlling the flow of fluid through the channel, so as to maintain the speed of rotation of the turbine within restricted limits in operation when the fluid is pumped through the drill string at a constant rate.

15. A hydraulic turbine according to Claim 14 in which the channel passes through the turbine rotor shaft communicating at one end with the turbine inlet and at the other end with the turbine outlet.

16. A hydraulic turbine according to either Claim 14 or Claim 15 in which said flow controlling means includes a valve located in the channel.

17. A well drilling system substantially as hereinbefore described with reference to Figures 1 and 2 or Figure 6 of the drawings accompanying the provisional specification.

18. A hydraulic turbine substantially as hereinbefore described with reference to Figures 1 and 2 or Figure 6 of the drawings accompanying the provisional specification.

19. A method of operating a well drilling system according to any one of Claims 1—13 and 17 in which the fluid circulation rate through the turbine is varied to keep the turbine speed within restricted limits.

20. A method according to Claim 19 in which the circulation rate through the drill string is maintained substantially constant.

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PROVISIONAL SPECIFICATION

Improvements in or relating to Well Drilling Systems and methods of Operating such Systems

We, N. V. De Bataafsche Petroleum Maatschappij, of 30 Carel van Bylandtlaan, The Hague, The Netherlands, a company organised under the laws of the Netherlands, do hereby declare this invention to be described in the following statement:—

The present invention relates to well drilling systems as may be used for example in drilling deep bore holes in the surface of the earth with the object of finding salt, sulphur, water, oil or the like.

The invention particularly relates to well

drilling systems of the type in which a tubular drill string, which in operation extends from the top of the bore hole to the bottom, is provided at its lower end with a hydraulic turbine for driving a rotary drill bit. The turbine is driven by fluid pumped down through the drill string, the fluid usually being what is known as a mud flush. Usually, too, the fluid is ejected through jet openings in the rotary drill bit, and is directed on to the cutting edges to cool them and to remove the cuttings, before circulating back to the surface.

When drilling with such a system through formations of varying hardness, the turbine speed will vary between wide limits, as a result of the varying resistance met by the bit. This may lead to stalling of the turbine when drilling in soft formations, or overspeeding in hard formations. In any case excessive deviations from the design speed of a turbine leads to inefficient operation. The turbine speed can be controlled manually by the driller at the top of the bore hole by varying the weight on the bit and/or the circulation rate of the mud flush which is supplied though the drill string to the turbine, in accordance with information, which is obtained about the turbine operation, such as the turbine speed or the pressure drop across the turbine. It is, however, often very difficult to communicate these measured variable from the bottom of the bore hole to the surface.

The same sort of difficulties are encountered when drilling with a system, for example that described in the provisional specification of our co-pending Patent Application No. 35904/54 (Serial No. 755,207), with reference to Figures 2 to 6 of the drawings, in which a ram device is interposed with the hydraulic turbine between the drill string and the rotary drill bit.

The ram device described comprises two hollow telescopic members, the inner one of which provides a channel for the passage of fluid from the drill string to the bit and which members have means for preventing relative rotational movement betwen them, one of said members being provided with one or more pistons, which co-operate with an equal number of cylinders formed on the other member, each piston dividing the cylinder with which it co-operates into two cylindrical spaces, the ram device also comprising means for connecting the cylindrical spaces located to one side of the pistons to a space in which there is a relatively high fluid pressure in operation, and means for connecting the cylindrical spaces located to the other side of the pistons to a space in which there is a relatively lower fluid pressure in operation. Such ram devices will be referred to in this specification as "ram devices of the type specified".

In the simplest case, the space in which there is relatively high fluid pressure is the interior of the inner telescopic member, and the space in which there is a lower fluid pressure is the exterior of the outer telescopic member (i.e. the borehole) so that the fluid pressures are those of the mud flush in the drill string and in the borehole respectively. The pressure difference applied to the pistons, and thus controlling the bit weight, is the difference of these two mud flush pressures, which is equal to the sum of the pressure drops across the turbine and the bit jet openings, both of which depend on the mud flush circulation rate. As explained in the aforesaid patent specification the bit weight determines the load on the turbine for a given hardness of the formation being drilled, and there is thus an interaction between the ram device and the turbine, which leads to an inter-relationship between the formation hardness, the turbine speed and the mud flush circulation rate. Here again it is important to restrict the variations in turbine speed in operation.

One may define in this connection, the sensitivity of a system as being the ratio between a range of formation hardness, and the corresponding turbine speed range when drilling through the said range of formation hardness. Thus, the larger the sensitivity of a system, the greater the range of formation hardness which may be drilled through without the turbine speed passing outside predetermined limits.

It is an object of the present invention to provide a method and means for increasing the sensitivity of a system.

According to the present invention in operating a well drilling system provided with a hydraulic turbine, or a combination of a hydraulic turbine and a ram device of the type specified, interposed between the drill string and a rotary drill bit, means being provided for circulating fluid through the drill string to the turbine, the circulation rate of the fluid through the drill string is kept constant and the circulation rate through the turbine is varied to keep the turbine speed within restricted limits, by passing part of the fluid through a channel by-passing the turbine.

The passage of fluid through the channel by-passing the turbine may be controlled by a valve responsive either to the pressure drop across the turbine or to the speed of rotation of the turbine. In either case the pressure drop across the turbine is preferably not permitted to rise above a predetermined value.

According to another aspect of the present invention a well drilling system provided with a hydraulic turbine, or a combination of a hydraulic turbine and a ram device of the type specified, for location near the lower end of a drill string, the turbine being arranged to drive a rotary drill bit and means being provided for circulating fluid at a constant rate through the drill string to the turbine, also includes a channel by-passing the turbine, and means for controlling the flow of fluid through the chan-

nel to keep the turbine speed within restricted limits.

Said means for controlling the flow of fluid through the channel by-passing the turbine may comprise a valve located in the channel, which valve is responsive either to the pressure drop across the turbine or to the speed of rotation of the turbine. In either case the valve preferably operates to prevent the pressure drop across the turbine rising above at a predetermined value.

Where the valve is responsive to the pressure drop across the turbine, it may comprise a valve member which is urged towards its closed position by a spring, and which in operation is subjected on one side to fluid at the pressure of the fluid at the turbine inlet and on the other side to fluid at the pressure of the fluid at the turbine outlet, in such a manner that the pressure difference acts in opposition to the spring.

Where the valve is responsive to the speed of the turbine it may comprise a valve member urged towards its closed position by a spring, and a centrifugal governor, which, as the speed of rotations of the turbine increases, causes a force to be exerted on the valve member in opposition to the spring. The centrifugal governor may for example control the rate of supply of fluid under pressure to a servo-mechanism, which, as the said rate increases, exerts an increasing force on the valve member in opposition to the spring.

In any of the above cases, the valve may be a sleeve valve.

The channel by-passing the turbine may comprise a channel through the turbine rotor shaft, which is thus hollow, the channel communicating at one end with the turbine inlet and at the other end with the turbine outlet, and containing the valve or other means for controlling the flow of fluid through it.

Further according to the present invention, there is provided a hydraulic turbine for location near the end of a drill string, the turbine being adapted for driving a rotary drill bit, and itself being capable of being driven by a supply of fluid pumped down through the drill string, the turbine being provided with a channel by-passing its blading, and means for controlling the flow of fluid through the channel, so as to maintain the speed of rotation of the turbine within restricted limits in operation, when the fluid is pumped through the drill string at a constant rate. The said channel may pass through the turbine rotor shaft communicating at one end with the turbine inlet and at the other end with the turbine outlet.

Examples of well drilling system in accordance with the present invention will now be described with reference to the accompanying drawings in which:—

Figure 1 is a cross-sectional view of the ram device/hydraulic turbine arrangement, comprising the by-pass valve as shown in detail in Figure 2;

Figure 3 shows the turbine pressure drop/turbine speed ($\Delta p/N$) curves for various values of mud circulation rate Q, passing through the turbine blading;

Figure 4 shows the torque/turbine speed (T/N) curves for various values of circulation rate Q, passing through the turbine blading;

Figure 5 shows the formation coefficient/turbine speed (C/N) curve from which curve the sensitivity of the ram device can be derived and

Figure 6 shows servo-motor operated by-pass valve.

Referring first to Figure 1, the turbine 1 comprises a stator housing 2 and a hollow rotor shaft 3, on which the turbine blading 4 is mounted. The rotor shaft 3 is supported by bearings 5 and 6, whilst the rotary drill bit 7, which may be of any suitable type, is connected to the lower end of the shaft 3.

Between the drill string 8 (which is kept stationary or is rotated only very slowly so as to prevent it from sticking) and the hydraulic turbine 1, there is interposed a ram device as described in the provisional specification of our co-pending Patent Application No. 35904/54 (Serial No. 755,207).

The ram device consists of a number of piston and cylinder combinations (for the sake of simplicity only two of these combinations are shown in the drawing but there may be more), the pistons 9 being connected to the inner hollow cylindrical member 10 and the cylinders 11 being formed on the outer hollow cylindrical member 12. Suitable sealing means (not shown) are provided between the pistons 9 and the interior surface of the outer member 12, and for sealing adjacent cylinders 11 from one another. The members 10 and 12 are telescopically arranged, a splined coupling 13 being provided to prevent any relative rotational movement between them. The cylindrical spaces located above each of the pistons 9 are connected to the interior of the inner telescopic member 10 by the ports 14 provided in the wall of the member 10, the ports 14 each being located just above one of the pistons 9, whilst the cylindrical spaces located below each of the pistons 9 are connected to the exterior of the outer telescopic member 12 by the ports 15 provided in the wall of the member 12, the ports 15 each being located at the lower end of the one of the cylinders 11 with which they are associated.

It will be clear that the difference existing between the pressure of the mud flush passing through the inner telescopic member 10, and the pressure of the mud flush in the borehole at the outside of the outer telescopic member 12 will be applied to all the pistons 9 and will exert on them a downwards force, which, in combination with the weight of the inner telescopic member 10, the turbine 1 and the bit

7, and the hydrodynamic force generated by the mud flush flowing through the turbine 1, will constitute the effective bit weight, i.e. the load on the bit.

Inside the hollow rotor shaft 3, there is provided a pressure-sensitive valve 16, which is urged towards the closed position by a compressed spring 17. The valve 16 will be closed for all conditions under which the pressure drop across the turbine 1 is below a predetermined value, and, whilst it is closed, the mud flush emerging from the inner telescopic member 10 will flow to the jet openings of the bit 7 through the turbine blading 4 and the ports 18 provided in the wall of the rotor shaft 3. When the pressure drop across the turbine rises above said predetermined value, the valve 16 will be opened, and part of the mud flush will by-pass the turbine blading 4 by flowing through the ports 19, the inside of the rotor shaft 3 and the valve 16. Thus, when the valve is open, the pressure drop across the turbine 1 will remain substantially constant, any increase in the rate of flow of mud flush down the drill string 8 resulting in an increased flow through the by-pass channel and not through the turbine blading 4.

It will be appreciated that in an alternative arrangement the valve 16 controlling the by-pass channel may be constituted by a piston-operated sleeve valve, one side of the piston being exposed to the pressure of the mud flush at the inlet of the turbine 3, and the other side to the pressure of the mud flush at the outlet of the turbine 3.

The relation between the turbine pressure drop Δp, the turbine speed N and the mud flush circulation rate Q the mud flush passing through the turbine blading 4 is shown by the curves in Figure 3. The turbine pressure drop/speed (Δp/N) curves are shown for various circulation rates Q. As indicated in the drawing, the circulation rate Q_1 is greater than Q_2, which in turn is greater than Q_3. The valve 16 is set to operate when the pressure difference across it is Δp_o, so that when starting to operate the turbine 3 with a constant mud flush circulation rate Q_1 through the drill string 8, the turbine speed increases from zero up to a speed N_1 at which the pressure drop across it has increased to Δp_o and the valve 16 is opened. At speeds higher than N_1, the pressure drop across the turbine will remain substantially equal to Δp_o, the circulation rate through the turbine 3 however dropping to Q_2 at speed N_2, the circulation rate through the by-pass valve 16 then being (Q_1-Q_2). Similarly at the turbine speed N_3, the corresponding circular rates are C_3 and (Q_1-Q_3).

The variations in the torque delivered by the turbine, as a result of by-passing part of the mud flush, may be seen in Figure 4, which shows the torque/speed (T/N) curves for the various circulation rates Q_1, Q_2 and Q_3 of the mud flush passing through the turbine blading 4.

It will be appreciated that when starting from the point O on the curve Q_1, the torque will be equal to T_1 when the turbine speed has increased to N_1. At this moment (see Figure 3) the pressure drop across the turbine is equal to Δp_o and the valve 16 opens. At the speed N_2, the circulation rate through the turbine will be Q_2 (Figure 3), which corresponds to a torque T_2, whilst at the speed N_3 the respective values are T_3 and Q_3.

It will be clear that, although only one curve (in this case Q_2) is shown as being representative of the circulation rates lying between the values Q_1 and Q_3, it is possible to draw intermediate curves for circulation rates lying between Q_1 and Q_3, in order to be able to plot the curve Δp_o in Figure 4 more exactly.

Figure 5 shows the formation coefficient/turbine speed (C/N) operation curve for a turbine driven bit, in which the bitweight W_o is a constant.

This curve W_o represents the operation in a case in which the bit-weight is kept constant either by adjustments made at the top of the borehole, or by a hydraulic ram device mounted at the lower end of the drill string 8.

The formation coefficient C can be derived from the following formula (see Oil and Gas Journal).

$$T=CdW$$

in which T=the torque in ft.lbs. (is *inter alia* a function of N)
d=the diameter in inches of the bit
W=the weight on the bit in lbs.
and C=the formation coefficient, which varies from 0.25 for hard formations, to 0.63 for soft formations.

As shown in Figure 5, the formation coefficient $C=\frac{T}{dW}$ is plotted against the turbine speed N, under the condition that the turbine pressure drop Δp remains substantially constant (see Figure 3).

The bit weight exerted by the hydraulic ram device can be calculated from the formula:—

$$W=A(\Delta p+kQ^2)$$

in which W=the weight on the bit
A=the effective area of the ram pistons
Δp=the pressure drop across the turbine
kQ^2=the pressure drop across the bit jet openings.

Since the area A, as well as the pressure drop $\Delta p(=\Delta p_o)$ and the pressure drop $kQ^2(=kQ_1^2)$ across the bit jet openings are constant factors, the bit weight W will under these conditions have a constant value W_c. Consequently the torque T will be proportional to the formation coefficient C, so that

the operational curve for constant bit weight W_o (Figure 5) may be easily derived from the T/N curve for constant pressure drop $\triangle p_o$ (Figure 4).

When drilling is started in a relatively soft formation having a formation coefficient C_1, and then proceeds into formations which become gradually harder, the coefficient increasing first to C_2 and then C_2 to C_3, the turbine speed will increase from N_1 to N_3 through N_2.

Comparing this arrangement with an arrangement having no by-pass, and in which the circulation rate Q is varied at the top of the borehole by the driller, so that the turbine pressure drop $\triangle p$ will be kept constant, the bit weight would be

$$W=A(\triangle p_o+kQ^2)$$

in which A as well as $\triangle p_o$ are constants, but in which kQ^2 varies from $kQ_1{}^2$ to $kQ_3{}^2$ as the turbine speed varies from N_1 to N_3. It will be obvious therefore that in this case the bit weight will not remain constant, but will decrease when the speed increases from N_1 to N_3, as the pressure drop across the bit jet openings at the same time decreases from $kQ_1{}^2$ to $kQ_3{}^2$.

In this case the formation coefficient will be influenced at higher speeds by the decrease of W, since the formation coefficient C is inversely proportional to the bit weight W, the operational curve W (Figure 5) for this system will be less steep than the operational curve W_o for a system according to the invention.

It will be clear too that the sensitivity of the system in this latter case i.e. $\frac{C_1-C_3{}^1}{N_3-N_1}$, is smaller than the sensitivity of the system according to the invention which is equal to $\frac{C_1-C_3}{N_3-N_1}$.

The object of the invention may be attained in another manner by inserting a speed-sensitive valve in the by-pass of the turbine. The valve which is operated by a centrifugal governor is set in such a way that starting from the speed N_1, an increasing quantity of mud flush is by-passed at increasing turbine speeds. It will be appreciated that this alternative arrangement will have an operational curve similar to the curve W_o shown in Figure 5.

In a preferred arrangement, the speed-sensitive by-pass valve is constituted by a servo-operated valve, the pressure fluid (mud) supply to the servomechanism being controlled by a centrifugal governor.

Such a by-pass valve is shown in Figure 6, mounted in the interior of the hollow rotor shaft 3.

The valve body 16, which is of the mushroom type (but which may also be of the sleeve type) is connected by a hollow stem 21 to the servo piston 22 which is arranged to slide in the servo cylinder 23. The valve body 16 is urged towards its closed position by the compressed spring 20. The cylindrical space below the piston 22 is connected to the turbine outlet space through the ports 27 and the interior of the hollow stem 21. A small bleeding duct 26 is provided through the piston 22. In the head of the cylinder 23, there is mounted a pilot valve 25 which is controlled by a centrifugal governor 24.

When the pilot valve 25 is fully closed at low speeds, the pressures in the spaces above and below the servo-piston 22 are equalised, owing to the presence of the duct 26, at a value equal to the fluid pressure at the turbine outlet.

When the turbine speed increases, the centrifugal governor 24 opens the pilot valve 25 and fluid from the space inside the shaft 3 enters the space in the servo cylinder 23 above the piston 22. Consequently the pressure in the space above the servo-piston 22 will rise until the rate of escape of fluid through the duct 26 is equal to the rate at which it enters through the pilot valve 25.

The force exerted by the pressure difference on the two sides of the piston 22 will displace the valve body 16 until the force as the piston 22 is in equilibrium with the opposing force due to the compression of the spring 20. The valve will then pass part of the mud flush, which would otherwise flow through the turbine blading 4.

The by-pass valve and the associated mechanism may be so designed that by their action the pressure drop across the turbine is kept substantially constant over the speed range in which it operates.

It will be appreciated that the influence of the pressure drop prevailing across the valve may be eliminated by designing it as a sleeve valve.

In general it is desirable to coat all the surfaces of the by-pass valve which are subjected to the erosive action of the mud flow with wear-resisting materials, and to design the by-pass valve in such a way that it is readily replaceable.

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Leamington Spa: Printed for Her Majesty's Stationery Office, by the Courier Press.—1956.
Published at the Patent Office, 25, Southampton Buildings, London, W.C.2, from which copies may be obtained.

762,749 PROVISIONAL SPECIFICATION

3 SHEETS

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SHEET 1

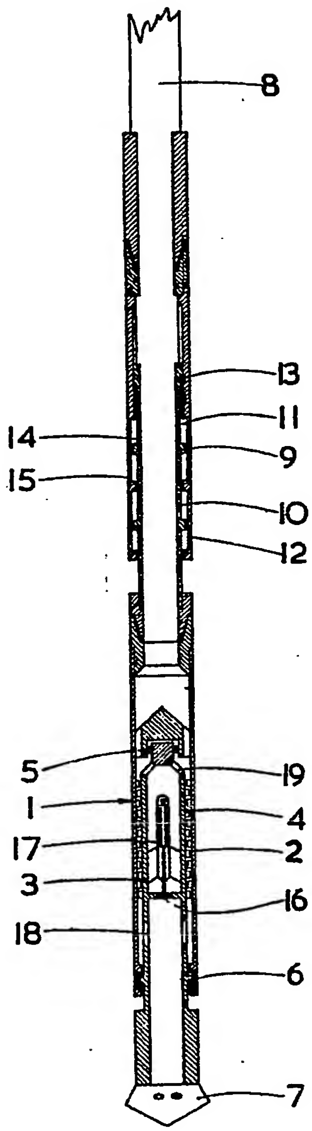

FIG. 1

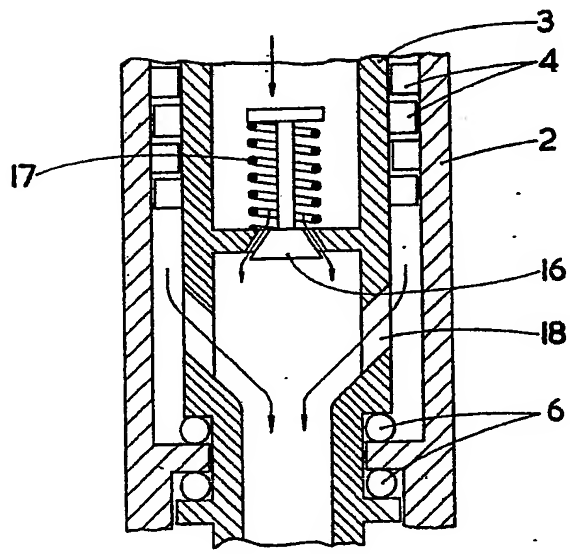

FIG. 2

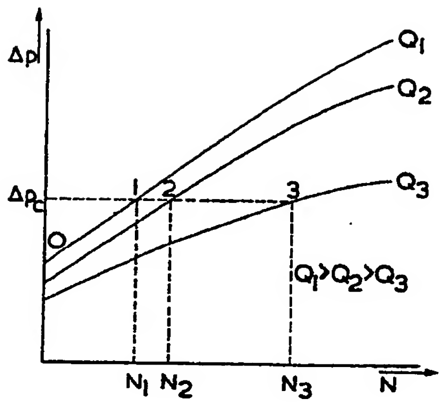

FIG.3

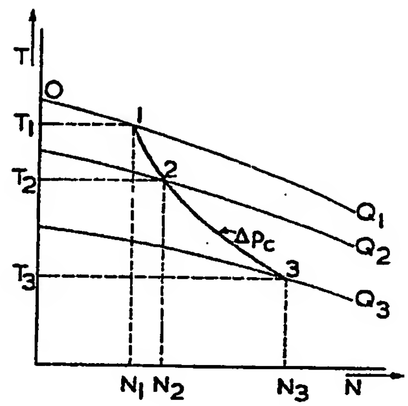

FIG.4

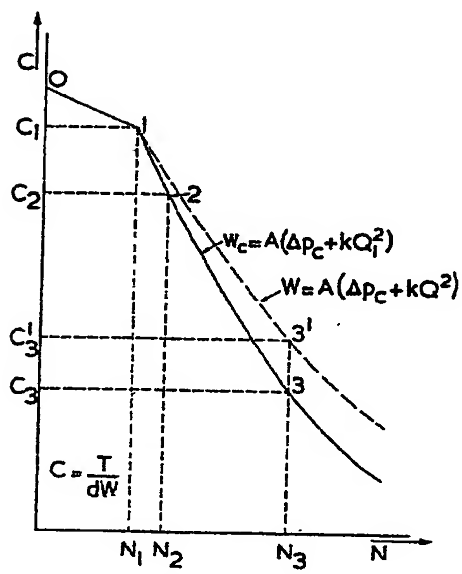

FIG.5

762,749 PROVISIONAL SPECIFICATION
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SHEETS 2 & 3

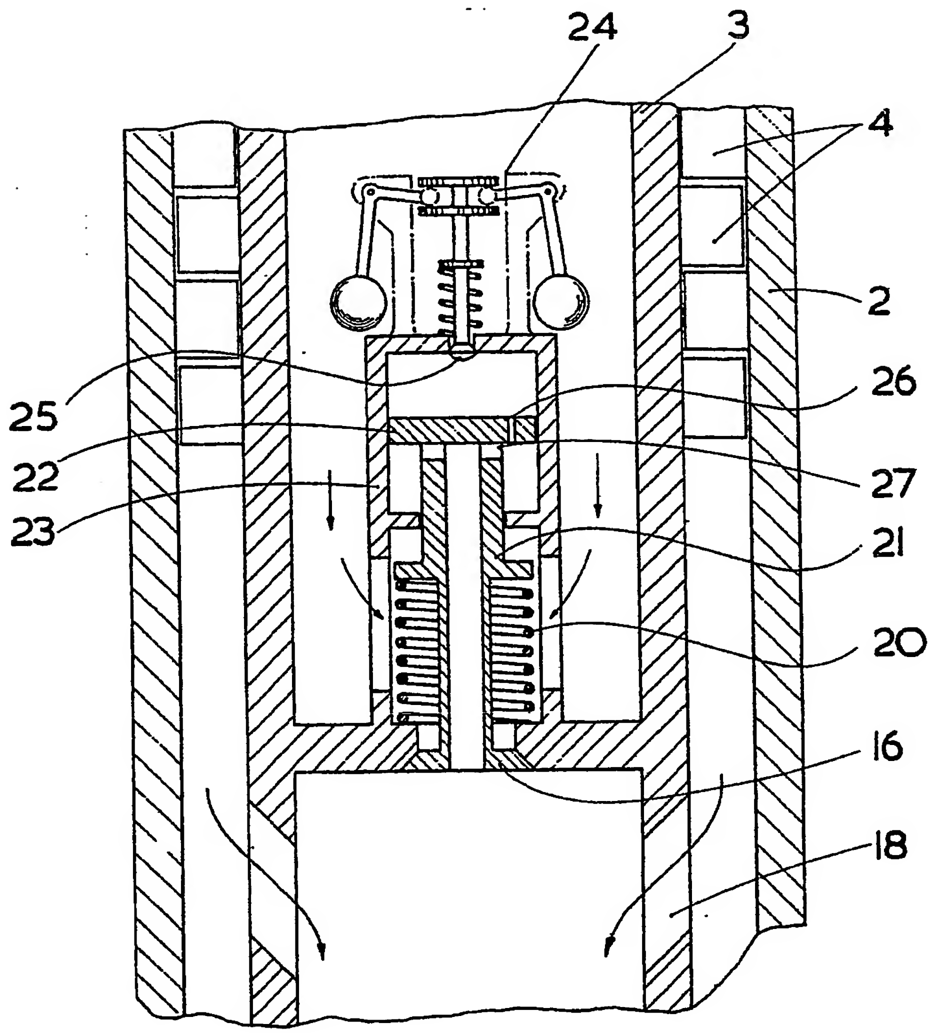

FIG. 6

762,749 PROVISIONAL SPECIFICATION
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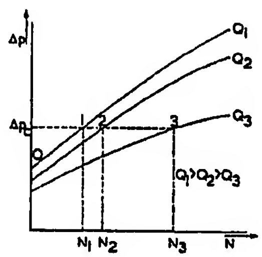

FIG.3

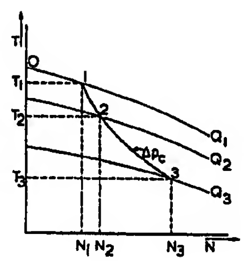

FIG.4

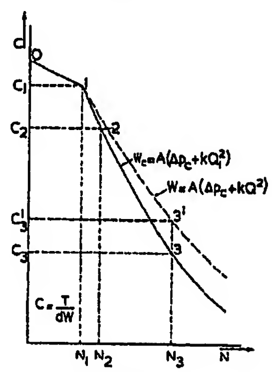

FIG.5

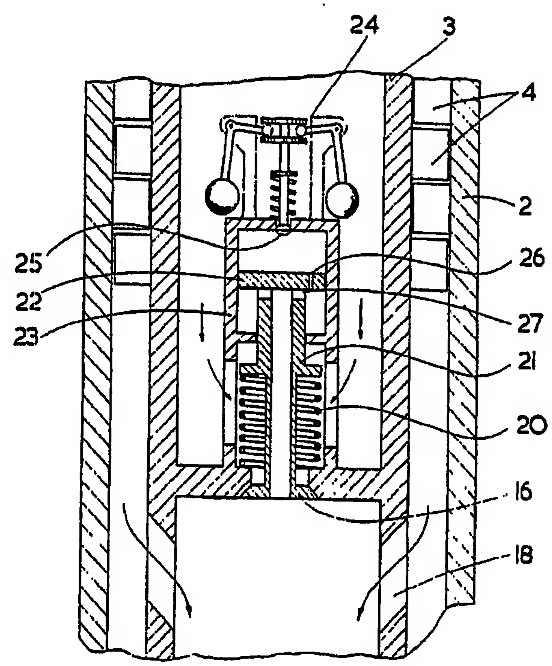

FIG. 6

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